

FIELD VALIDATION OF THE ASHRAE TRANSFER FUNCTION METHOD
FOR CALCULATING COOLING LOAD

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ABSTRACT

The American Society of Heating, Refrigerating and Air Conditioning Engineers (ASHRAE) Transfer Function Method (TFM) was validated using two identical wood frame residential-type structures, each containing an east and west-facing room. Each room had a calibrated, thermostatically controlled window air conditioning unit and two south-facing windows. The study included a parametric analysis of the thermostat setpoint and fenestration load effect on space heat extraction rate and cooling energy consumption. Some discrepancies withstanding, the transfer function method predicted the hourly heat extraction rates quite well. The principal discrepancies appeared to be the difference in daily curve amplitude and a phase-like shift of one to two hours. The heat storage capacity of the unoccupied test buildings was less than predicted by the TFM model. Accuracy of the transfer function coefficients to model the roof-ceiling combination was questionable due to the small attic air space which was not accurately described in the ASHRAE table of coefficients.

INTRODUCTION

Cooling loads account for the major portion of the residential utility bill during summer in the South. Sizing of air-conditioning equipment to handle the loads is accomplished with one of many standard procedures such as the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) Cooling Load Temperature Difference (CLTD) method or the Air Conditioning Contractors of America (ACCA) Manual J method. These analytical procedures estimate the maximum cooling load by using steady state type equations based on fundamental laws of conduction heat transfer. These methods are obviously approximate, as such important factors as heat storage and outside surface orientation are only marginally included in the calculations. A more precise method is the ASHRAE Transfer Function Method (TFM). This model utilizes transfer functions, computed from rigorous calculations involving the governing equations of heat transfer applied to typical residential structures, to calculate space cooling loads and heat extraction rates on an hour-by-hour basis.

There are many other rigorous hour-by-hour cooling load programs developed by the various

government agencies and universities. Gadgil et al. (1) demonstrated that the majority of these large mainframe programs were equivalent. In fact, many of the program algorithms were based on ASHRAE procedures.

An hour-by-hour profile of the cooling load is of interest as an energy consumption predictor since instantaneous heat gains are closely matched with the instantaneous cooling load and space heat extraction rate. In addition to giving the maximum cooling load which the air-conditioner must meet, such a cooling load profile identifies the time of day at which it occurs, and the method allows estimation of daily energy consumption to provide the cooling.

Before describing the Transfer Function Method, some definitions are relevant at this point.

1. Space heat gain is the instantaneous rate at which heat enters or is generated in an enclosed space.
2. The space cooling load is the rate at which heat must be removed to maintain a constant space temperature.
3. The heat extraction rate is the rate of heat removal by the air-conditioning system from the space, and it equals the space cooling load only if the space temperature remains constant. In a normal daily cycle temperature changes slowly, so that at any time the heat extraction rate can be taken as equal to the cooling rate.

The ASHRAE Transfer Function Method (TFM) was developed by Stephenson and Mitalas (2). The model utilizes transfer functions, also called weighing factors or thermal response factors, computed from rigorous calculations involving the governing equations of transient heat transfer applied to typical residential structures. In developing the Transfer Function Method, Stephenson and Mitalas considered the governing equations for a surface heat balance of the six surfaces of a room and those governing the transient heat conduction within the building components adjoining those surfaces. Initial determination of the coefficients involved a matrix formulation. However, a more recent formulation of the coefficients utilized Z-transforms (Stephenson and Mitalas, (3); Mitalas, (4)).

A useful validation of the ASHRAE model for residential interests would involve a relatively small, tight structure that is still representative of residential structures and one in which the more uncertain heat gain components, such as air infiltration and internal sensible and latent loads, may be neglected entirely in the modelling. Such a structure was available at Louisiana State University, School of Architecture. In fact, two 300 ft² identical, fully instrumented wood-frame buildings, with two test rooms each, were used.

OBJECTIVES

The following were the project objectives:

1. Validate the ASHRAE Transfer Function Method of predicting the hourly heat extraction by the air-conditioning system.
2. Test certain aspects of the model by
 - a. changing the thermostat setpoint temperature.
 - b. changing the fenestration load.

COMPUTER MODEL OF THE TRANSFER FUNCTION METHOD

Figure 1 shows a schematic relation of the transfer function method for predicting cooling load, space extraction rate and ultimately indoor space temperature. It is essentially the logic diagram for the computer model written to utilize the transfer function method. Heat gain calculations were based on conduction through exterior walls, floor, and the roof-ceiling combination. Except in the case of the floor, these calculations were based on sol-air temperatures. Solar and conduction heat gain through windows was the only other component of heat gain considered. Infiltration was assumed to be negligible because all windows and air-conditioners were tightly sealed. Also, all doors contained magnetic strips. In addition, the rooms were always unoccupied and contained no internal heat or moisture. Complete model information and validation results were reported by Quille (5).

The test buildings had 1/4 slope pitched roofs. Due to the absence of tabulated transfer coefficients for pitched roofs in the ASHRAE handbooks, (6,7), the roofs were modelled as being horizontal. The ASHRAE equations for conduction heat gain were used to calculate the heat gain through exterior walls and the roof-ceiling combination. The ASHRAE coefficients associated with the designation of "light" construction was used, as opposed to "medium" or "heavy" construction.

The transfer function coefficients were chosen from ASHRAE tables to match as closely as possible the actual structure. This was relatively easy for the walls because there were 103 different wall sections in the ASHRAE literature from which to choose transfer coefficients. But, as pointed out above, the roof was of low slope while ASHRAE coefficients are given for horizontal, and the

transfer coefficients used represented less attic insulation than was the actual case.

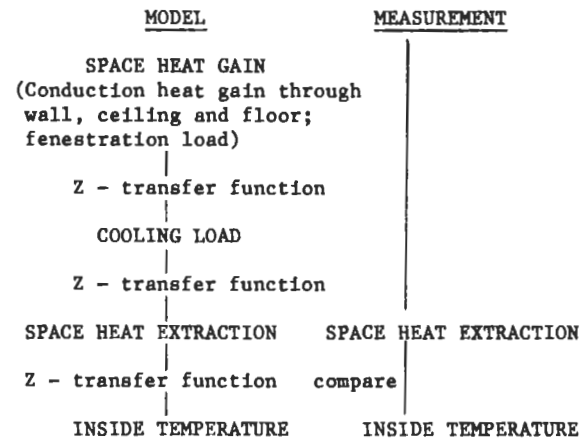


Fig. 1. Relation of transfer function to field measurements of room cooling load and heat extraction rate.

TEST BUILDINGS

The validation procedure consisted of measuring hourly temperatures and cooling energy consumption in two identical, skid-mounted, test houses, each containing an east and west-oriented room with an instrumentation room between. The buildings were of light wood-frame construction, and the walls were insulated with R-11 of fiberglass insulation, the attic and floor with R-19 of the same insulation. Figure 2 gives details. Each room initially contained two windows. However, since the fenestration area to floor area ratio was considered to be much larger than usual residential construction, one window in each room was covered, sealed, and insulated to wall levels. This left a fenestration area to floor area ratio of 11.5. Unlike most residential structures, however, there was no roof overhang or exterior or interior shading. This absence of shading was actually advantageous when it came to modelling solar gains. Each room had a window-type air-conditioning unit (Heat Controller, Inc., Jackson, MI) of 4,600 Btu/hr capacity. Thermostats for each air-conditioner were Accustats, Model TMS-AH-23 with precision (0.5°F) sensors, mounted on the interior walls of a room.

INSTRUMENTATION

The pertinent measurements to the analysis were air-conditioning power usage and inside temperature. The latter was accomplished with a shielded thermocouple (Type J) located close to the interior instrumentation room wall. This point was chosen as it was away from direct air currents set up by the air-conditioner and free from direct solar gains. All sensors were polled every seven seconds and the data were recorded and then averaged over hourly intervals on the hour with a data logger (Doric Scientific, Inc., Digitrend 235, San Diego, CA). Other parameters measured were wall and ceiling

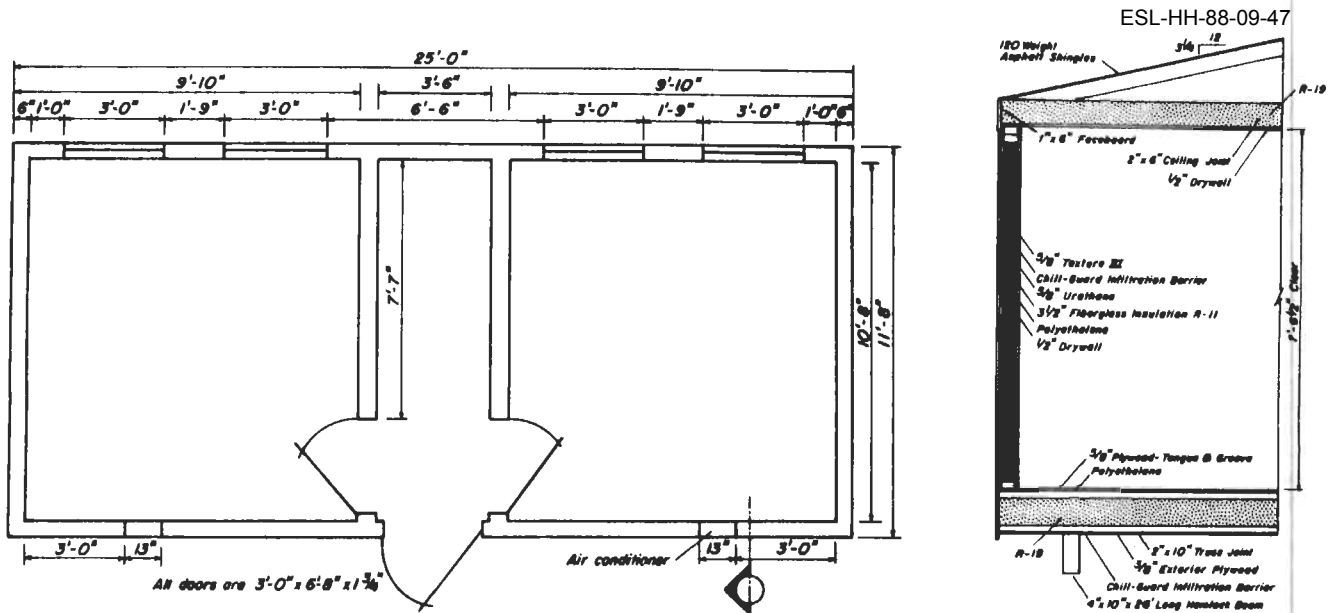


Fig. 2. Floor plan and wall section of test building.

surface temperatures, temperature at the thermostat location, solar radiation and attic temperatures.

Solar radiation data were taken with a pyranometer (Epply Laboratory, Inc., Model PSP, Newport, R.I.) mounted on the vertical surface of the south building.

Recent research (8) has shown that attic insulation temperatures may be higher than attic air temperatures during midday hours. Thus, roof surface, attic air, and attic insulation temperatures at a depth of 1 inch were measured.

Electric energy consumed by an air-conditioner was converted to millivolts by a watt transducer. Power consumption was calibrated with a watt-hour meter (Ohio Semitronics Inc., Model WH3, Columbus, Ohio).

COP DETERMINATION FOR THE AIR-CONDITIONING UNITS

The coefficient of performance (COP) is defined as a ratio of the total heat removed by a cooling unit divided by the electrical energy input. This ratio had to be determined for each individual room air-conditioning unit, since the cooling unit power consumption would be measured during experiments. For COP determination, a metal duct was fabricated and placed on the air discharge of each room cooling unit. Simultaneously with the temperature measurements, air speed was determined with a hot wire anemometer (TSI, Model 1610-12, St. Paul, MN). This instrument had previously been calibrated in a wind tunnel. The COP values determined from these measurements are reported in Table 1.

Room	E-N	W-N	E-S	W-S
Run 1	1.26	1.21	1.24	1.37
Run 2	1.55	1.45	1.53	1.40

Table 1. Coefficient of Performance (COP) for the Four Room Air-Conditioning Units.

The first run values were measured during a five minute on-time and the second run during a ten minute on-time. Normal on-time was closer to five minutes during air-conditioner operation.

EXPERIMENTAL PROCEDURE

Three tests were run to compare measured hourly cooling energy to that predicted by the transfer method model. The test period was August and September, 1987.

1. The first experiment was a base case in that all four rooms had identical thermostat setpoints (78°F) and identical windows (double pane). Thus, in addition to comparison with the ASHRAE model predictions, inherent room differences were determined. The test was repeated later with all single pane windows.

2. The second experiment involved a study of the influence of thermostat setpoint on heat extraction and cooling load. The north building was operated at 78°F and the south operated at 74°F indoor temperature.

3. This was a window study. Rooms E-S and W-S of the south building had double pane windows which was compared to single pane windows in the north building.

DETERMINATION OF INHERENT DIFFERENCES IN TEST ROOMS

The first analysis examined the inherent differences that could be discerned between the four test rooms with the same thermostat settings and same window treatment. As the computer model assumed cloudless skies, only those days which were the sunniest and cloudless were chosen, August 20, 22, 24, and 28, for all double pane windows, and Sept. 10, 12, and 13, for all single pane windows.

Data collected and model predictions are presented in Figure 3 and Table 2. Very erratic

operation of the air-conditioning unit in west room W-S was noticed during the tests with double pane windows. Eventual failure of the thermostat required that it be replaced. Therefore, data for this test room were recorded but not used in subsequent analyses.

Referring to Figure 3 for the east rooms, the principal discrepancies between the heat extraction rates predicted by the TFM model and measured appear to be the difference in amplitude and the phase-like shift of one to two hours. This shift, which begins to appear soon after daybreak, suggests that the heat storage capacity of the buildings was less than the model. This would also help explain the difference in peak loads. Room E-N (double pane) peak load was 7% larger than E-S and 41% higher than the TFM model peak, Table 2. Room E-S had a 32% higher than model peak for double pane and 18% with single pane. The difference between east rooms was only 7% for double pane and 18% for single pane. The west room W-S had a 11% (single pane) higher peak load than the model predicted, and room W-N had a 10% lower peak. Room difference was 19% single pane. These results were used to normalize the data in the following experiments.

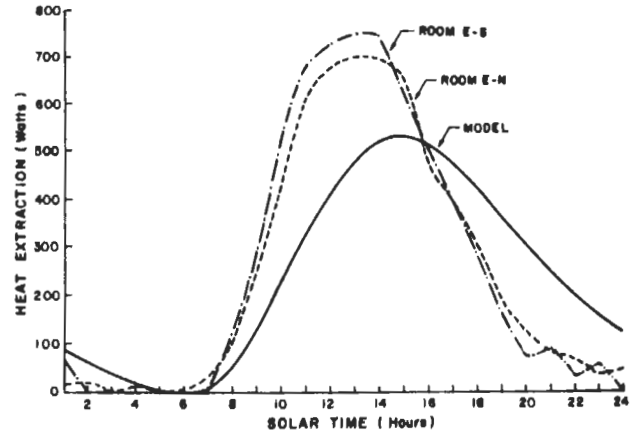


Fig. 3. Experimental and predicted heat extraction rates for rooms E-N and E-S. Double pane windows.

Table 2. Comparison of Measured Peak Heat Extraction and Cooling Energy Consumption to TFM Model.

	Set	Window	Room				Model-E	Room				
			East-N	Diff.	East-S	Diff.		W-N	Diff.	W-S	Diff.	Model-W
BASE STUDY												
Peak watts	78 ^o	Double	749	7%	699	32%	530	671	-	*	-	550
	78 ^o	Double	749	-	-	41%	530	671	-	-	22%	550
Peak watts	78 ^o	Single	658	18%	798	18%	679	625	19%	772	11%	694
	78 ^o	Single	658	-	-	-3%	679	625	-	-	-10%	694
Energy watt-hr	78 ^o	Double	5894	3%	6069	7%	5655	5649	-	*	-	5660
	78 ^o	Double	5894	-	-	4%	5655	5649	-	-	-1%	5660
Energy watt-hr	78 ^o	Single	4834	10%	5393	-22%	6902	4770	18%	5837	15%	6901
	78 ^o	Single	4834	-	-	-30%	6902	4770	-	-	-31%	6901
THERMOSTAT SET POINT												
Peak watts	78 ^o	Double	520	-	-	11%	583	605	-	-	1%	600
Difference			-	47%	-	-	8%	-	-	-	-	8%
Peak watts	74 ^o	Double	-	-	769	22%	630	-	-	*	-	647
Energy watt-hr	78 ^o	Double	3983	-	-	37%	6063	4321	-	-	-29%	6064
Difference			-	49%	-	-	18%	-	-	-	-	18%
Energy watt-hr	74 ^o	Double	-	-	5936	17%	7164	-	-	*	-	7164
WINDOW STUDY												
Peak watts	78 ^o	Single	845	-	-	21%	699	821	-	-	15%	715
Difference			-	13%	-	-	9%	-	7%	-	-	8%
Peak watts	78 ^o	Double	-	-	732	15%	639	-	-	770	18%	655
Energy watt-hr	78 ^o	Single	6187	-	-	12%	7066	6500	-	-	8%	7067
Difference			-	12%	-	-	8%	-	10%	-	-	8%
Energy watt-hr	78 ^o	Double	-	-	5430	17%	6529	-	-	5934	-9%	6529

*Data void due to faulty thermostat.

Besides heat storage, other factors might explain discrepancies between the measured and predicted data. A flat roof was assumed in the modelling of the roof-ceiling combination. Also, for the ceiling-attic-roof combination, a ceiling U-value larger than the actual one was used in the model. However, as McQuiston (9) noted, partly due to the attic air space resistance used in the roof-ceiling heat gain transfer function coefficients, their applicability to this building component is limited.

Over the period from 1100 hours to 1800 hours, for the two east rooms, the average measured heat extraction rate was 12.3% higher than predicted, and for one west room W-N 11.4% higher.

Close agreement in total daily cooling energy were in evidence. Measured energy for the east rooms were 7% and 4% higher than the predicted by the model. Correspondingly, for the west room, the measured energy was 1% higher than predicted with double pane but 31% below with single pane.

INFLUENCE OF THERMOSTAT SETPOINT

The effects of lowering the room thermostat setpoint were investigated. To achieve this, the north building rooms had their thermostats set to 78°F, and rooms of the south building had their setpoints set to 74°F. Figure 4 provides graphic results.

A shift in time of the peak load is seen again. The difference in peak load is smaller than in the previous test, because the slightly less sunny weather conditions during this experiment. Peak heat extraction rates for Rooms E-N (78°F) and E-S (74°F) were 11% and 22% smaller than predicted, respectively. The corresponding figures for Room W-N was 1% larger than predicted. Total daily heat extraction energy varied considerably, probably because of ideal sunny conditions.

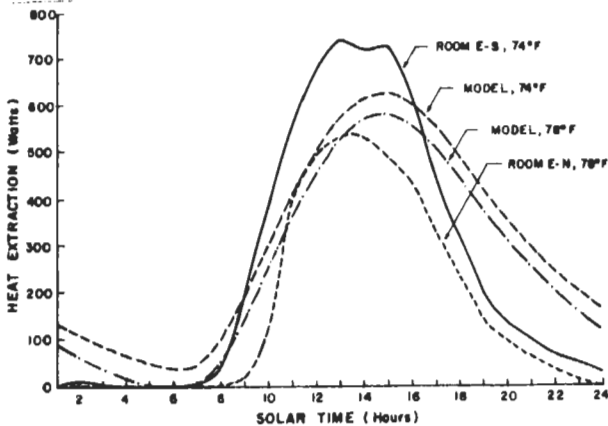


Fig. 4. Influence of thermostat setpoint on measured and predicted heat extraction rates for rooms E-N and E-S. Double pane windows.

The model predicted an 8% increase in peak heat extraction for an east room upon going from 78°F

thermostat setpoint to 74°F, Table 2. The measured hourly increase was 47% after adjustment based upon the room differences found in the base study. Energy consumption increase for lowering the thermostat setpoint to 74°F was 18% predicted by the model, while the measured daily increase was 49% for the west room.

INFLUENCE OF FENESTRATION LOAD

The third experiment examined room heat extraction rates for two different window types, single pane windows in the north building and double pane windows in the south. The experimental data covered the days September 14-15, and 17, Figures 5-6.

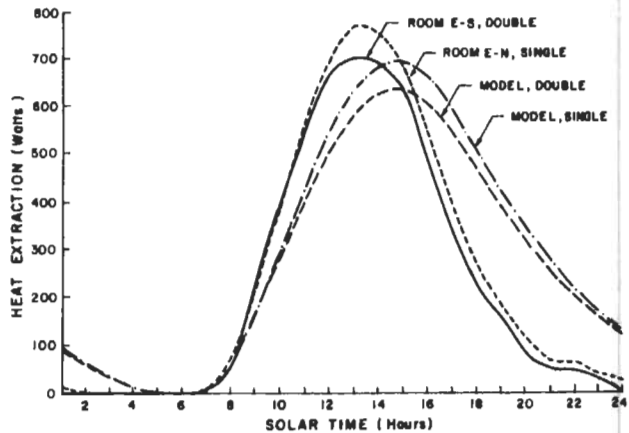


Fig. 5. Influence of fenestration load on measured and predicted heat extraction rates for rooms E-N and E-S.

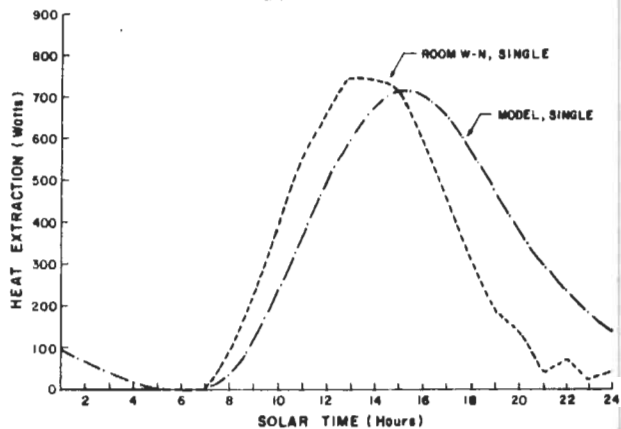


Fig. 6. Influence of fenestration load on measured and predicted heat extraction rates for room W-N.

Overall, this test displayed the closest agreement between measured and predicted data. Measured peak loads were still marginally higher than modelled, 21% for single pane and 15% for double pane, east rooms. For the west rooms, the

data yielded a 15% larger measured peak for single pane and 18% larger peak for double pane.

The main interest was the energy savings to be made by reducing the window load. The TFM model predicted an 8% savings for double pane over single pane. After data adjustment in light of the room differences found in the base study, the measured savings for double pane over single pane was 12% in east rooms and 10% in west rooms.

SUMMARY

The ASHRAE transfer function method of calculating heat extraction rates was validated using two identical test buildings. Three test runs were performed:

1. The four test rooms were operated with the same indoor thermostat setting of 78°F and with the same windows to discern inherent room differences.
2. The second test was a study of thermostat setpoint with two rooms set at 78°F and the other two at 74°F.
3. The third study compared single pane to double pane windows.

A pronounced shift in phase and amplitude between the measured and predicted heat extraction rates curves in the early days (August) of the experiment suggested that heat storage and peak solar conditions were not modelled accurately then. However, in the later days of the experiment (September) these differences diminished. Since the prevailing external temperatures remained more or less the same over the period August 20 to September 17, this suggests that the model is very sensitive to solar altitude.

Some discrepancies withstanding, the transfer function method predicted the hourly heat extraction rates quite well. The principal discrepancies appeared to be the difference in daily curve amplitude and a phase-like shift of one to two hours. The heat storage capacity of the unoccupied test buildings was less than predicted by the TFM model. Accuracy of the transfer function coefficients to model the roof-ceiling combination was questionable due to the small attic air space which was not accurately described in the ASHRAE table of coefficients.

A parametric study was done on the influence of thermostat setpoint on room heat extraction. The setpoint value seems to have a greater effect than predicted by the model. The model predicted an 8% increase in peak heat extraction upon going from a thermostat setpoint of 78°F to 74°F. The measured increase was 47%. The predicted increase in total heat energy for the thermostat setpoint variation was 18%. The measured increase was 49%.

The ASHRAE model predicted an 8% energy savings for the double pane windows over single pane windows. The measured savings were 12% for east

rooms and 10% for the west rooms. Finally, predicted hourly temperatures tended to exceed the measured ones by one degree. This was relatively close agreement considering the difficulty in precisely modelling thermostat operation.

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